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ABSTRACT (Maximum 200 words)

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The original engineering background to this work was the need to understand the behaviour and failure of gear tooth contacts used in demanding aerospace applications such as aircraft engine and helicopter gearboxes. Scuffing tests, designed to measure the performance of a particular combination of lubricant and gear steel, are usually impractical on full-scale machinery, while tests using smaller gears are expensive. Furthermore, it is difficult to measure quantities of interest for research purposes, such as friction and bulk temperature. Fundamental research investigations of scuffing under controlled conditions are therefore more conveniently carried out through the use of disc machines.

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Title: The influence of surface treatments on micropitting and scuffing

Investigators: M P Alanou, R W Snidle and H P Evans

Final report

CARDIFF UNIVERSITY
UK

1- Introduction

This is the final report of the contract issued on 1 May 2001; this contract was a follow-on of contracts N68171-98-M-5294 and N68171-99-M-6457. Hence this report is based on an update of actions described in previous interim reports either covered by contracts N68171-98-M-5294, N68171-99-M-6457 or by the current contract. The focus of the current contract was on the scuffing performance of an ultra hard thin coating optimised for gearing applications.

Scuffing tests were performed with steel disks prepared as follows:

- Ground (Ra = 0.4 μm ± 0.05 μm) case carburised steel with ultra hard thin coating optimised for gearing applications performed at sliding speeds of 7 m/s, 16 m/s and 20 m/s.
- Superfinished (Ra < 0.1 μm) case carburised steel with ultra hard thin coating optimised for gearing applications performed at sliding speeds of 7 m/s, 16 m/s and 20 m/s.

2- The scuffing test bench

2-1 Brief description of the test bench

The scuffing test bench as used for this test programme has already been described in detail in the final report for contract N68171-99-M-6457. Hence only key features of the facility will be described here. The original engineering background to this work was the need to understand the behaviour and failure of gear tooth contacts used in demanding aerospace applications such as aircraft engine and helicopter gearboxes. Scuffing tests, designed to measure the performance of a particular combination of lubricant and gear steel, are usually impracticable on full-scale machinery, while tests using smaller gears are expensive. Furthermore, it is difficult to measure quantities of interest for research purposes, such as friction and bulk temperature. Fundamental research investigations of scuffing under controlled conditions are therefore more conveniently carried out through the use of disc machines.

Apart from the high speed/high temperature capability other special features are the use of crowned discs to give a self-aligning contact, and axial finish of the disc surfaces, which reproduces the orientation of finish found on gears. An outline of the specification of the test rig is as follows:

- Sliding speed up to 23 m/s (75.5 ft/s)
- Maximum Hertzian contact pressure up to 1.7 GPa (247,000 lbf/in^2)
- Oil feed temperature up to 200 deg C
- Lubricant typically Mobil Jet II synthetic

The layout of the test head and its drive arrangement is shown in Figure 1. Each test shaft is supported on a two-row spherical roller bearing at one end, and a cylindrical roller bearing at the other end. The spherical roller bearing provides high radial load capacity together with axial location, and the cylindrical roller bearing provides radial support without axial restraint. The shafts are of EN36 steel (equivalent to AMS 9310), case carburised and hardened to 680 Vickers Hardness Number (30 kg). The

shafts were carefully finish ground on dead centres. An interference fit was provided between the discs and shafts (minimum 0.015 mm, maximum 0.025 mm on diameter) and the discs are pressed on to the shafts using a hydraulic press. After testing, the discs are pressed off the shafts and the shafts are re-fitted with new discs. The shafts are provided with central and radial feed holes for thermocouple leads to allow measurement of the bulk temperature of the discs. A general view of the machine is shown in Figure 2.

Toothed belts and pulleys drive the two test shafts from the two outputs from a splitter gearbox. A toothed belt drives the input shaft of the gearbox and pulleys from a 18.5 kW TASC variable speed drive motor. The maximum speed of the TASC unit is 2800 rpm so speed-increasing pulley ratios are used to achieve the highest speeds required. The drive was designed to operate the machine with a maximum test shaft speed of 12,000 rpm which gives a disc surface speed of 47.9 m/s. By using different combinations of pulleys on the output shafts of the splitter gearbox and drive shafts of the test head it was possible to obtain test shaft speed ratios from 1 (pure rolling) to almost 5. The surfaces of both discs moved in the same direction relative to their contact, but at different speeds, depending on the gear ratio selected.

The machine is designed to impose a maximum load of 4 kN at the contact which, for the geometry chosen (i.e. 76.2 mm diameter discs, each with 304.8 mm crown radius) gives a maximum Hertzian contact pressure of about 1.7 GPa (247,000 lbf/in^2). The load is applied by means of a hydraulic ram through a bell-crank and push rod mechanism. The slower speed disc is mounted in a swinging arm bearing housing as shown and the end of the push rod in contact with this arm made contact through a crossed knife edge so that the load was applied at an accurately pre-determined point. In this way any tendency to twist the loading arm (which would have moved the contact band between the two discs) was minimised. The line of action of the loading is designed to pass through the contact between the discs. A proprietary electrical resistance load cell mounted on the bell crank provides a direct measurement of the load. A rapid reduction of the load is required when scuffing failure between the discs is detected so as to leave some of the "run-in" surface of the discs for profile measurements. This is achieved by providing a manually operated quick-release valve

in the hydraulic supply to the ram, which rapidly dumps the system pressure back to the tank.

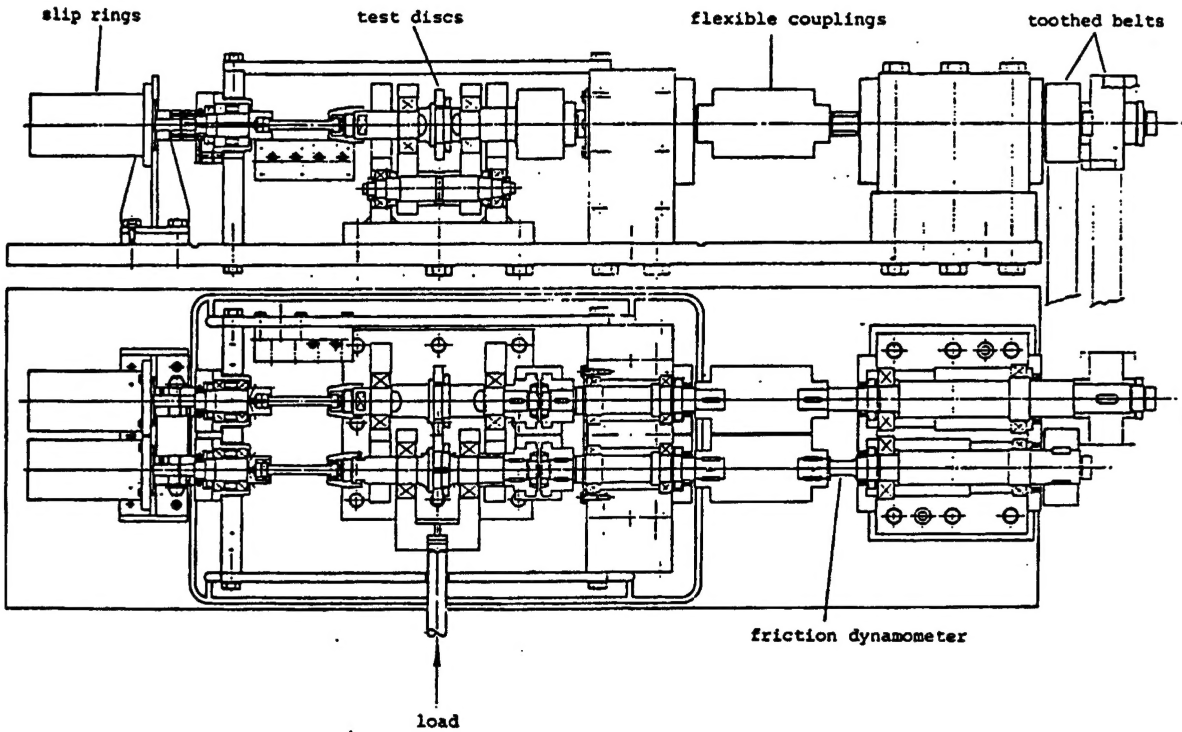

Figure 1 Section and plan drawing of the test head and drive arrangement

The test oil is circulated to the test discs and support bearings from an electrically heated tank which is designed for heating oil at temperatures up to 300 deg C. The maximum power input to the tank is 4.6 kW. The contents of the tank are stirred rapidly to prevent the development of hot spots. A hydraulic pressure filter with a steel mesh element rated at 1 μm is used in the test oil supply line. The filter body is fitted with special high temperature seals, and all supply and drainage pipes to and from the test head are of stainless steel.

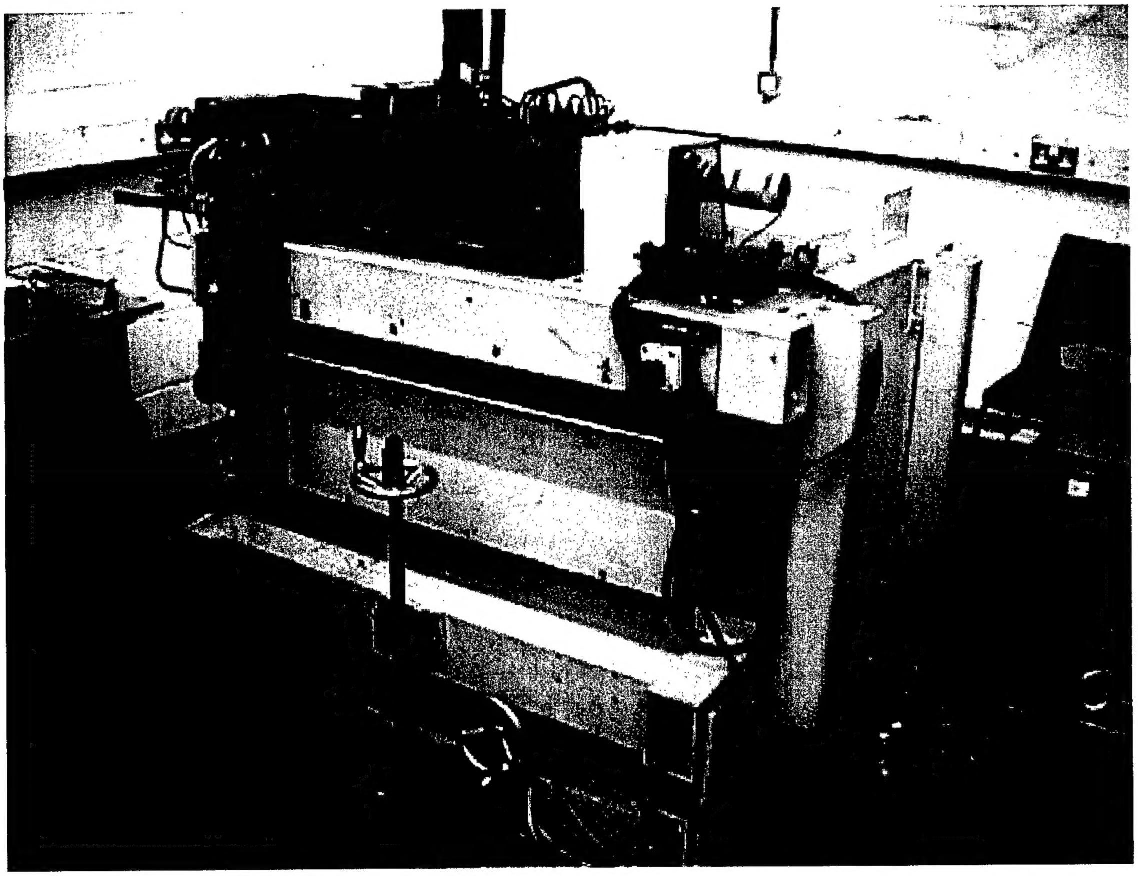

Figure 2 Overall view of the scuffing rig

The bulk temperature of the surfaces in the inlet to the lubricated contact that is of key importance in elastohydrodynamic lubrication is measured by embedding thermocouples just beneath the surfaces of both discs. The leads from these thermocouples are connected through silver/graphite slip rings to a continuous recorder. The slip rings are of a special type for use with thermocouples. The thermocouples were of the iron/constantan type insulated with PTFE. The thermocouple junction is made in a small copper ferrule of 2.0 mm diameter which is a close fit in a hole provided in the test discs. The ferrule is pressed and expanded into the hole with a hollow punch. This provides a reliable mechanical anchorage of the junction and also ensures good thermal contact between the junction and the steel disc. The thermocouple leads are routed through the hollow test shafts and short drive shafts, having flexible couplings at both ends, to the slip rings which are mounted outside the test head as shown in Figure 1.

The tangential friction force between the discs is measured by monitoring the torque in the shaft driving (or, strictly, braking) the slower speed test shaft. A suitable dynamometer shaft is fitted with electrical resistance strain gauges, which are connected to an electronic unit through a third silver/graphite slip ring unit. The strain gauge units were bonded to the dynamometer shaft, and temperature sensitive compensating elements were incorporated to eliminate thermal drift of the unit. The output from the electronic unit is also continuously recorded on the chart recorder.

As for the test discs themselves, it is recognised that the appropriate direction of their surface finish is axial, because this corresponds to the direction of finish on real gears which is transverse to the rolling and sliding direction. A method of axially grinding the discs by a generation process has been developed at Cardiff.
It was also decided to "crown" the discs so as to give a self-aligning configuration and avoid edge contact and consequent high contact pressures at the edges of the discs. Crowning also avoids the difficult problem of ensuring a uniform distribution of load on the discs in the axial direction. The choice of crowning radius depends upon the width of the discs, the heaviest load to be applied and the required tolerance to misalignment and errors in the centring of the crown on the discs. The discs were chosen to be nominally 10 mm wide and a crowning radius (on both discs) of 304.8 mm (12 inches) was selected. The principal radius of relative curvature at the contact between the two discs is therefore 19.05 mm in the circumferential direction and 152.4 mm in the axial direction. This gives a ratio of principal radii of curvature of 8.0 and a ratio of the axes of the Hertzian elastic contact ellipse of 3.91, with the minor axis of the contact aligned in the circumferential (entraining/sliding) direction.

2-2 Progress of a typical scuffing test

The load applied to the contact between the discs is carefully controlled and known as the “standard” loading sequence. A standard loading regime is adopted in which the load is increased at intervals of 3 minutes until either scuffing failure is detected or the load limit of the machine is reached before a scuff occurs. The increment by which the load is increased at each stage is chosen to give a constant increment in the corresponding dry contact (Hertzian) pressure for a nominally smooth surface. The

load stages and the corresponding loads and maximum Hertzian pressures are given in Table 1 below.

Table 1 Standard loading sequence used in scuffing tests

Load Stage	Time (min)	Load (N)	Max Hertz Pressure (GPa)
1	0	180	0.6
2	3	290	0.7
3	6	430	0.8
4	9	620	0.9
5	12	850	1.0
6	15	1120	1.1
7	18	1460	1.2
8	21	1850	1.3
9	24	2320	1.4
10	27	2850	1.5
11	30	3450	1.6
12	33	4150	1.7

To conduct a test the machine is assembled with the discs and shafts in place. The test oil is first heated to the desired oil feed temperature by switching on the test oil heaters. During this heating period the test oil is circulated through the test head so that static thermal equilibrium of the machine is attained. The test oil is heated to 100 deg C and the gearbox oil to 45 deg C. A light load is then applied through the hydraulic ram to bring the discs into light static contact. This load is typically 100 N.

The main drive motor for the machine is then started and the speed gradually increased manually up to the pre-determined speed for the test. The speed is checked by the digital read out of the rpm counters attached to the two shafts. When the required speed is set the machine is allowed to run for a short period of time, typically three minutes, to allow the bulk temperatures of the two discs to stabilise. The first

load of the standard loading sequence as described earlier is then applied with a manually operated proportional valve.

The progress of the test in terms of temperature and friction measurements is continuously recorded on a chart recorder (see Figure 3). As described earlier a torque meter mounted on the slow shaft measures the friction.

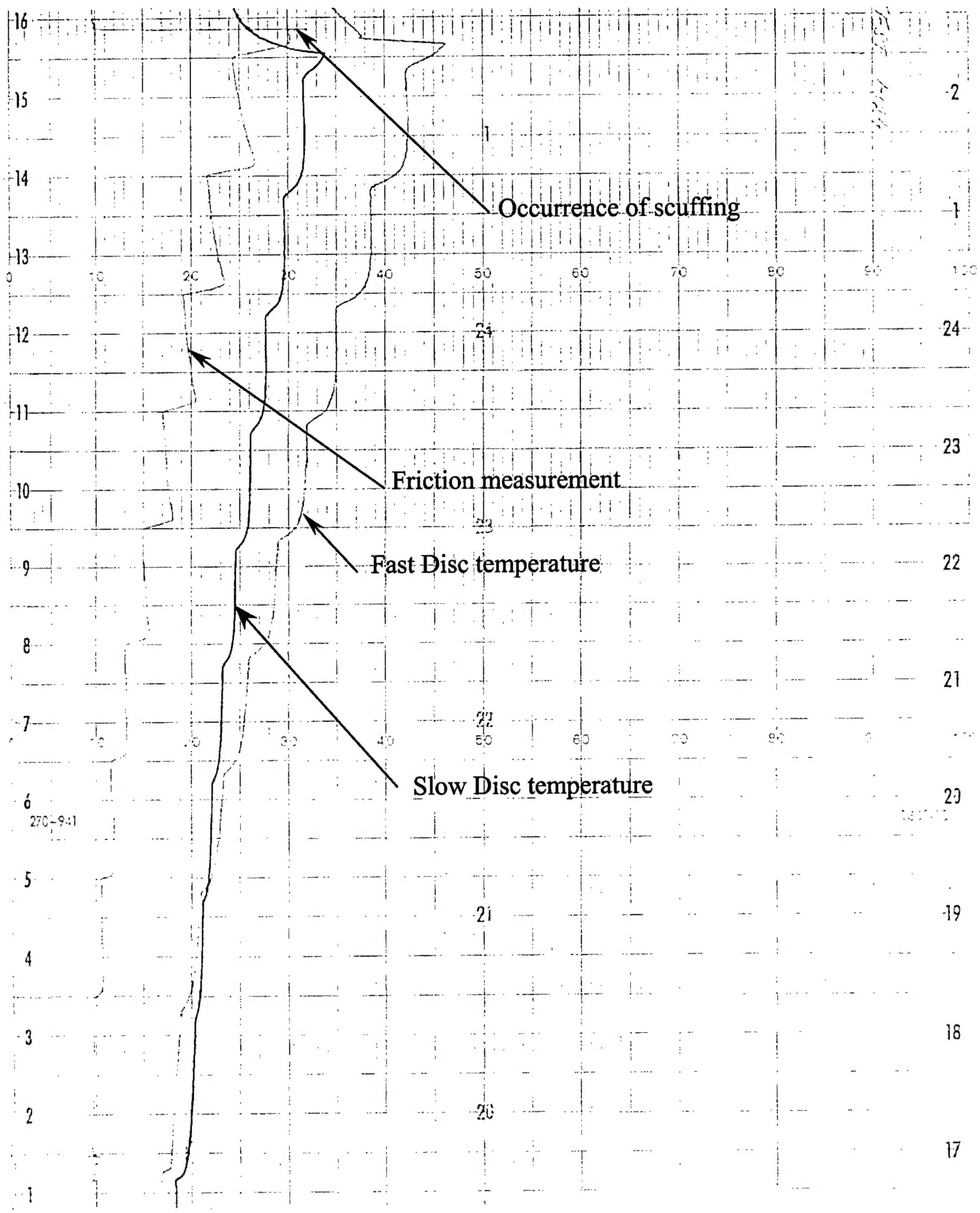

Figure 3 A typical test trace from the chart recorder

A feature of the tests is that at lower loads an increase of load produces a sudden rise in both friction and temperatures, which then level off to a steady value. At the higher load stages, however, there is a distinct tendency for the friction and temperature to first rise after an increase in load and then to fall back to lower values. This is attributed to a running in effect during which, it is suggested the surface conditions improve, so reducing friction and the generation of heat.

As the test approached the loads at which a scuff is expected a careful watch is kept on the chart recorder for signs of the scuff occurring. Scuffing produces an unmistakable sharp rise in friction accompanied by corresponding rapid increases in the temperature of both discs as shown in Figure 3. At this point the load is relieved and the drive motor turned off so the friction returns to zero and the temperatures of the discs fall rapidly.

When the machine has cooled down the covers are removed and the discs inspected for confirmation that scuffing has occurred. They are then removed from the machine for further inspection and surface analysis.

3- Scuffing test results

3-1 Introduction

From the award of the first contract, a comprehensive scuffing test campaign has been completed. Configurations tested for this particular test programme covered by the current contract were as follows:

- Ground (Ra = 0.4 μm ± 0.05 μm) case carburised steel with ultra hard thin coating optimised for gearing applications performed at sliding speeds of 7 m/s, 16 m/s and 20 m/s.
- Superfinished (Ra <0.1 μm) case carburised steel with ultra hard thin coating optimised for gearing applications performed at sliding speeds of 7 m/s, 16 m/s and 20 m/s.

All these tests have been performed using the standard applied load regime on the discs i.e. up to 12 load stages corresponding to increments of 0.1 GPa of equivalent Hertzian dry contact pressure. When scuffing is detected (as described above), the test is stopped.

3-2 Entraining and Sliding Speeds

The aim of the experimental work is to investigate conditions typical of the contacts found in existing engineering applications particularly helicopter gearboxes. Sliding speeds are typically limited to 16 m/s but in future designs much higher speeds are contemplated, possibly as high as 25 m/s. When the rig was designed, the aim was therefore to achieve this target of sliding speed if possible.

The choice of rolling speed, given a desired sliding speed, depends upon the slide/roll ratio which is defined as the ratio of relative sliding speed to mean entraining speed.

If two surfaces are rolling and sliding together with surface velocities relative to their conjunction of U_1 and U_2 then the sliding speed is:

$$U_s = U_1 - U_2$$

and the mean entraining, or rolling, velocity is: $U_r = \frac{U_1 + U_2}{2}$

The slide/roll ratio is therefore: $\frac{U_s}{U_r} = 2\frac{(U_1 - U_2)}{(U_1 + U_2)}$

In a gear tooth contact this ratio varies during the meshing cycle. When contact occurs at the pitch point the sliding velocity is zero so the slide/roll ratio is also zero. The maximum slide/roll ratio usually occurs at the gear tooth tips. The maximum values of slide/roll ratio calculated for typical gas turbine engine gears range from about 0.4 to 0.8.

In a two disc machine with discs of equal diameter, and in which both surfaces move in the same direction relative to the contact, as they would if connected by a single pair of gears, the slide/roll ratio may be expressed in terms of the ratio, G of the shaft speeds as:

$$\frac{U_s}{U_r}=2\frac{(G-1)}{(G+1)}$$

where G is the "gear ratio" = $U_1/U_2 \geq 1$.

Hence slide/roll ratios obtainable in the machine vary from 0 (Gear ratio of 1) to 2.00 (Gear ratio of ∞ with one disc stationary), the latter condition may be described as "simple sliding".

The highest gear ratio currently attainable in the two-disc machine used in this work is 4.24, which gives a slide/roll ratio of 1.24

The choice of slide/roll ratio to be used in a disc machine in order to provide an exact simulation of gear tooth action is not straightforward because the entraining action of the inlet region of an elastohydrodynamic contact depends upon the geometrical conformity of the surfaces as well as their mean speed. For line contacts the geometrical conformity is described in terms of the radius of relative curvature of the two surfaces. In the disc machine used in this work the radius of relative curvature in the entraining direction is 19.05 mm. In spur gears the radius of relative curvature between tooth contacts depends, in general, on the position considered in the meshing cycle. An approximate indication of the radius of relative curvature, which applies when contact occurs at the pitch point, is given by:

$$R=\frac{r_1 \cdot r_2}{(r_1+r_2)}\sin\theta$$

where θ is the pressure angle of the gear teeth and r_1 and r_2 are the pitch radii of the two gears in mesh. If we take the special case of equal gears so $r_1 = r_2 = r$ and a typical pressure angle of 20 degrees, we see that for R = 19.05 mm we obtain r = 111.4 mm. The disc machine contact is therefore geometrically equivalent (in the entraining direction) to the contact (at the pitch point) between a pair of equal, 20 degrees pressure angle gears, each of pitch diameter 222.8 mm.

As far as the test procedure is concerned, a typical assessment for a particular configuration (i.e. a particular combination of heat treatment, surface finish and/or coating) would consist of nine tests run at three different sliding speeds respectively 7, 16 and 20 m/s. When availability of test pieces is limited, the chosen sliding speed would be 16 m/s, which is considered as the "standard" sliding speed. For that particular speed, a complete performance comparison is possible for all configurations.

3-3 Reference test: Case Carburised with Ground finish

In order to allow for useful comparison it is important to recall scuffing results performed on the same test bench using case carburised steel (equivalent to AMS 9310). These tests were performed by Patching et al. [1] (M tests) and others were performed during contract N68171-98-M-5294. The main results for various sliding speeds are summarised in Table 2.

The main conclusions for these tests with ground surfaces can be summarised as follows:

- At lower loads an increase of load produces a rapid rise in both friction and temperature that then level off to a steady values. At the higher load stages, however, there is a tendency for the friction and bulk temperatures of the disks to rise and then fall back to lower values. This is attributed to a "running in" effect during which, it is suggested, the surface conditions improve, so reducing friction and the generation of heat.

- Scuffing invariably occurs at the edges of the contact area (running track) and this is attributed to a side leakage effect, i.e. oil leaking through the roughness valley features in the axial direction near the edge of the contact.

Table 2 Summary of results obtained by Patching *et al.* [1] and during contract N68171-98-M-5294 on Case Carburised, Ground (un-coated) steel

Test n°	M23	M24	M25	18	M14	M2	M16	17	M3	23
Peripheral Velocity of Fast shaft (m/s)	9.15	9.15	9.15	9.15	13.09	17.01	20.95	20.95	24.28	26.18
Peripheral Velocity of Slow shaft (m/s)	2.15	2.15	2.15	2.15	3.09	4.01	4.94	4.94	5.73	6.17
Mean Entraining Velocity (m/s)	5.65	5.65	5.65	5.65	8.09	10.5	12.95	12.95	15.01	16.18
Sliding Velocity (m/s)	7.00	7.00	7.00	7.00	10.00	13.00	16.00	16.00	18.50	20.00
Scuffing Load (N)	1860	2321	2840	2399	2320	2320	1850	1921	1880	1510
Maximum Bulk Temperature of Fast Disc (°C)	175	185	186	153	226	227	215	206	271	210
Maximum Bulk Temperature of Slow Disc (°C)	130	135	136	129	159	164	152	155	206	160
Mean Bulk Temperature of Discs (°C)	152.5	160	161	141	192.5	195.5	183.5	180.5	238.5	185.0
Maximum Peak Hertzian Contact Pressure (GPa)	1.30	1.40	1.50	1.42	1.40	1.40	1.30	1.32	1.31	1.21
Traction Coefficient at Scuffing Load	0.040	0.034	0.031	0.033	0.031	0.030	0.029	0.023	0.040	0.020

Figure 4 is a chart showing the scuffing load for each test versus the sliding speed. and includes a least squares fit to the ten tests.

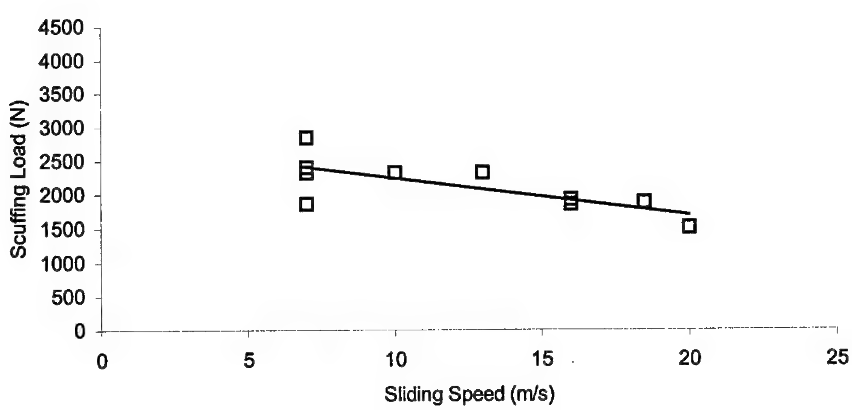

Figure 4 Experimentally determined scuffing loads as a function of sliding velocity for the ten ground surface (un-coated) tests (least squares fit also shown)

Circumferential profilometer traces from a disc used in Patching's work are shown in Figure 5.

Traces (a) and (b) are from the fast disc. Trace (a) was taken from the disc before the test was run and trace (b) was taken from an un-scuffed part of the running track after the test was completed. These two profiles show significant modification to the surface caused by the running in effect.

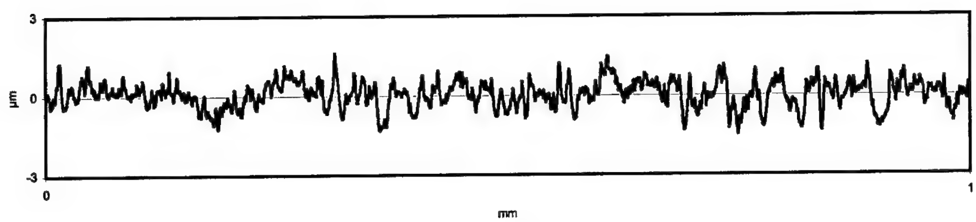

Figure 5a **Un-run surface profile from a Case Carburised Ground Disc used in a scuffing test**

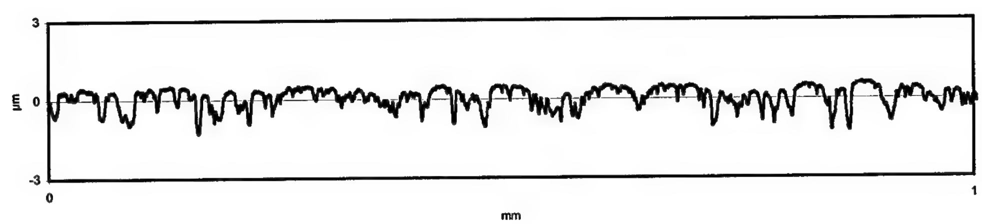

Figure 5b **Run-in but un-scuffed part of surface profile from a Case Carburised Ground Disc used in a scuffing test**

3-4 Test results for ground case carburised steel with an optimised thin hard coating

3-4-1 Test results with the ground surface

Nine tests were carried out with samples made of case carburised steel (AMS 9310) coated with the optimised thin hard coating. The results of these tests are summarised in Table 3.

Table 3 Summary of test results from ground case carburised steel discs coated with an optimised thin hard coating

Test n°	59	60	61	65	66	67	69	72	73
Peripheral Velocity of Fast Shaft (m/s)	9.15	9.15	9.15	20.95	20.95	20.95	26.18	26.18	26.18
Peripheral Velocity of Slow Shaft (m/s)	2.15	2.15	2.15	4.94	4.94	4.94	6.17	6.17	6.17
Mean Entraining Velocity (m/s)	5.65	5.65	5.65	12.95	12.95	12.95	16.18	16.18	16.18
Sliding Velocity (m/s)	7.00	7.00	7.00	16.00	16.00	16.00	20.00	20.00	20.00
Scuffing Load (N)	4118†	4136†	4123†	4189†	4190†	4176†	4198†	4303†	4320†
Maximum Bulk Temperature of Fast Disc (°C)	124	121	124	169	167	172	182	175	213
Maximum Bulk Temperature of Slow Disc (°C)	111	115	117	141	133	132	146	148	152
Mean Bulk Temperature of Discs (°C)	117	118	121	155	150	152	164	161	183
Maximum Peak Hertzian Contact Pressure (GPa)	1.70	1.70	1.70	1.71	1.71	1.70	1.71	1.72	1.72
Traction Coefficient at Scuffing Load	0.010	0.010	0.010	0.007	mal func.	0.006	mal func.	mal func.	0.007

†: no scuffing occurred; the load corresponds to the maximum testing load

Figure 6 summarises the performance observed in terms of load versus sliding speed. Scuffing loads as shown on the graph correspond effectively to the maximum load as encountered during tests as for all tests, *no scuffing was observed even at the highest possible contact pressure.* In all such tests this maximum contact pressure was maintained for about 30 minutes. This indicates a remarkable performance of this particular coating across the whole sliding speed range.

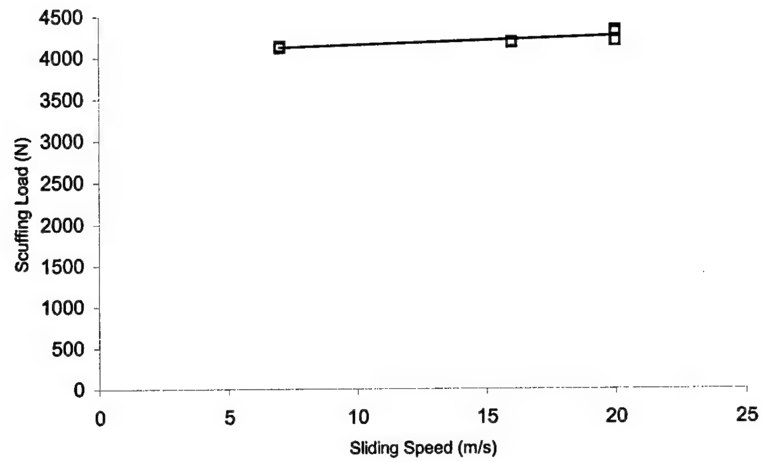

Figure 6 **Experimentally determined scuffing loads as a function of sliding velocity for the nine case carburised + optimised thin hard coating tests (least square fit also shown).**

It may be noted that running-in took place in these tests. An interesting feature for this particular configuration is that the onset of running-in seems to occur at a fairly low load starting, for example, as early as the second load stage (corresponding Hertz pressure: 0.7 GPa). The running-in is indicated by a sharp drop in friction force occurring immediately after having applied the new load level. The drop in friction force is typically of the order of 20 % from the maximum reached during the load stage. This behaviour could be attributed to the change of metallurgical structure of Diamond Like Carbon (DLC) coatings taking place under heavy contact loading combined with sliding [2] [3]. This phase transformation is supposed to be responsible for the change from a diamond like structure to a graphite like structure. This phenomenon tends to result in a superfinished-like type of surface appearance towards

the end of the test as illustrated in Figure 7 which shows surface profiles taken before and after testing. Note, however, that a number of deep valleys remain.

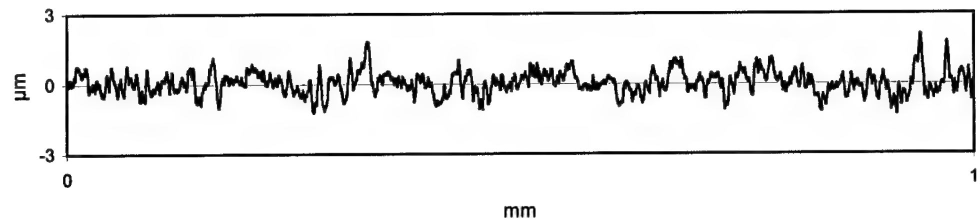

Figure 7 (a) **Surface profile from a case carburised steel disk coated with an optimised thin hard coating measured along the circumferential direction prior to testing.**

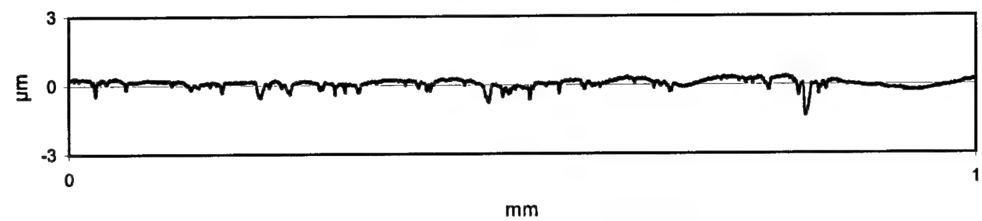

Figure 7 (b) **Surface profile from a case carburised steel disk coated with an optimised thin hard coating measured along the circumferential direction after test (test run at 16 m/s sliding speed).**

Overall, the appearance of the test samples after test exhibit a fairly similar pattern irrespective of the testing conditions i.e. for all sliding speeds. Hence, as illustrated in Figure 8, there is a central band across the width of the disc corresponding to the width of the contact at the maximum testing load. This band has a polished appearance due to running-in as illustrated in Figure 7. The colour of the band is darker than the un-run surface. There is a narrower band located in the middle of the contact in the area of maximum contact pressure. This band is typically about 2mm wide and appears to reveal the bare substrate especially at the tips of the roughness asperities.

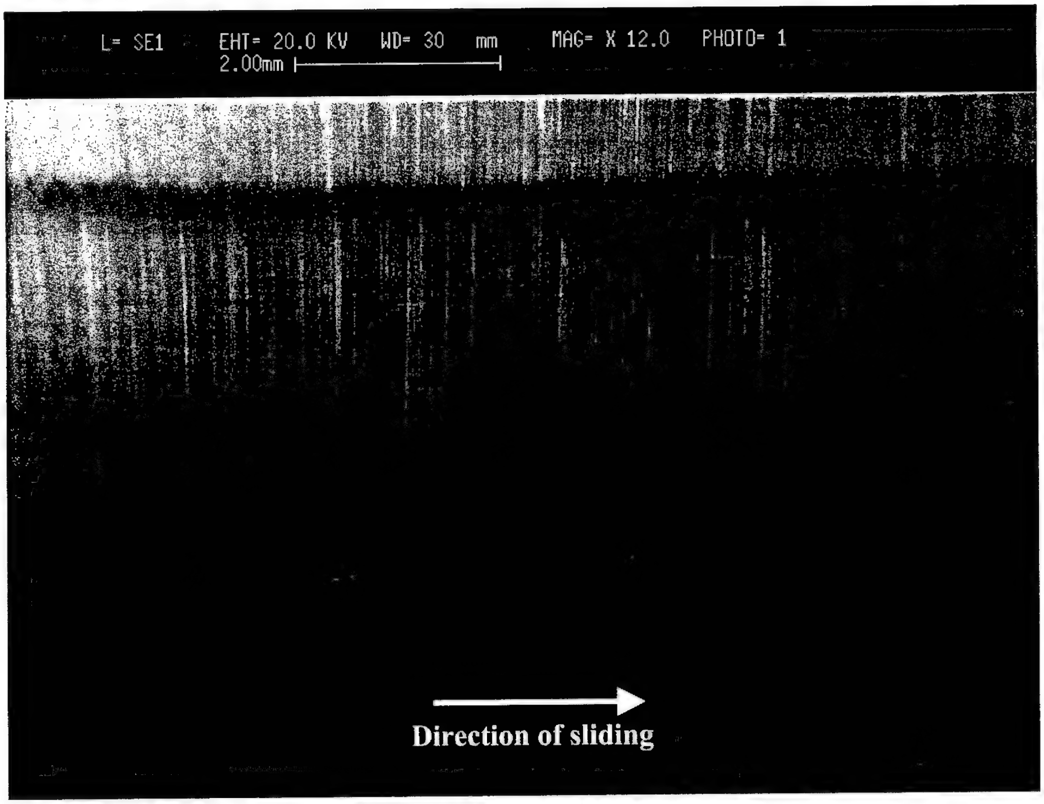

Figure 8 **SEM photograph of a ground coated sample after a test run at 16 m/s sliding speed, showing the central band corresponding to the contact width.**

Jiang and Arnell [4] have studied the effect of substrate roughness on DLC coatings and consider a critical maximum contact pressure that triggers the onset of a wear mode characterised by crack formation and flaking. The suggested critical contact pressure is about 3.7 GPa for the DLC coatings studied. A contact simulation using a solver such as the one developed at Cardiff University [5] could be employed to establish the pressures at asperity level. Alanou et al. [2] have already predicted micro-asperity pressures of this magnitude in elastohydrodynamic simulations using surfaces of similar roughness. In this investigation [2] the surfaces were case carburised samples coated with a metal-doped diamond like carbon (Me-DLC) coating. Surface deterioration in the middle of the contact is particularly evident for tests performed at the highest sliding speed (20 m/s). For example in test n° 73, the middle of the track of the "slow" disk, shown in Figure 9, shows delamination of the coating although scuffing (as indicated by a sharp increase in friction and

temperature) was not experienced after running for 30 minutes at the maximum load (1.7 GPa). Had the maximum load been maintained for a much longer time, it is suspected that scuffing would have occurred in the middle of the running track rather than at the edges which is the usual behaviour with ground surfaces.

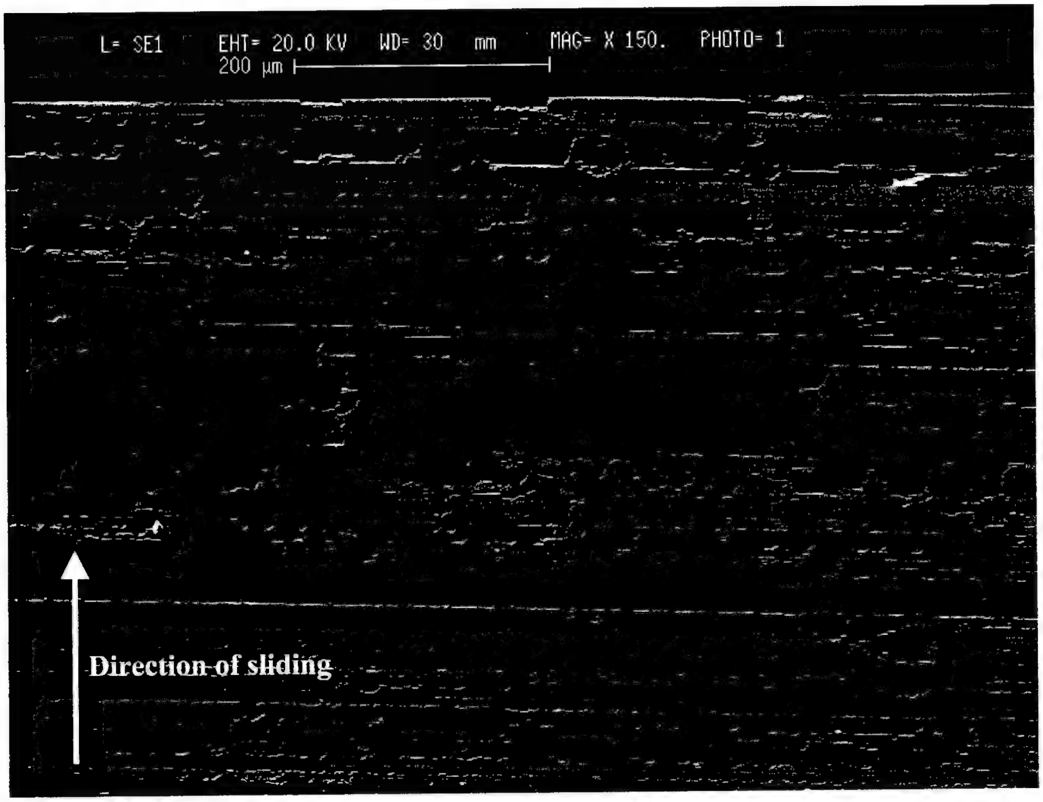

Figure 9 **SEM photograph of a ground and coated sample after a test run at 20 m/s sliding speed (test n°73), showing delamination of the coating at the middle of the contact area.**

Regarding mean bulk temperatures of the discs at the scuffing or maximum load, an interesting comparison can be drawn with the baseline uncoated reference as illustrated in Figure 10. The coated samples operate with significantly lower temperatures and there is less scatter in the temperatures measured. The reduction in mean bulk temperature is about 50 deg C over the whole range of sliding speeds considered. and this is consistent with much lower friction recorded during the tests.

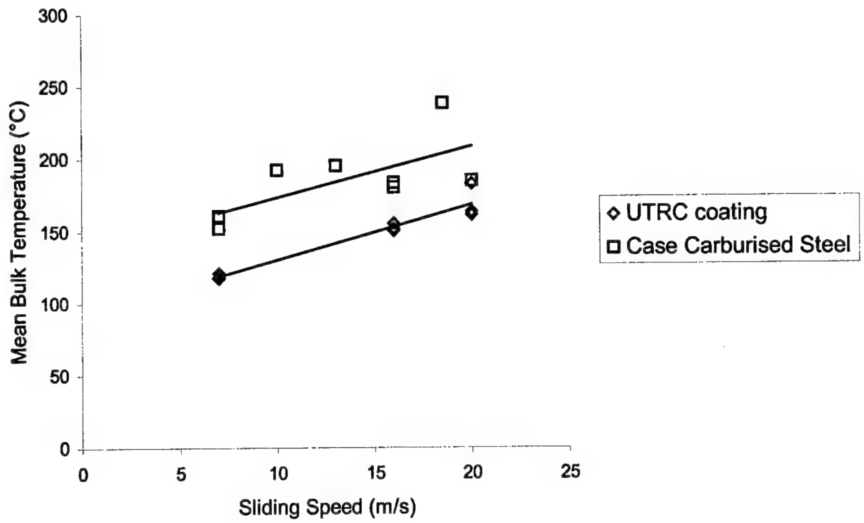

Figure 10 **Experimentally determined maximum mean bulk temperatures as a function of sliding velocity for the ground case carburised with optimised thin hard coating coated surface (UTRC coating), and the reference un-coated case carburised surface tests (least square fits also shown).**

As mentioned earlier no scuffing (as defined by a sharp increase in friction and temperature) has been experienced at the maximum load attainable in the test rig (maximum Hertzian contact pressure of 1.7 GPa), even for the highest sliding speeds. This compares favourably with the performance of the baseline of simply ground samples as illustrated in Figure 11.

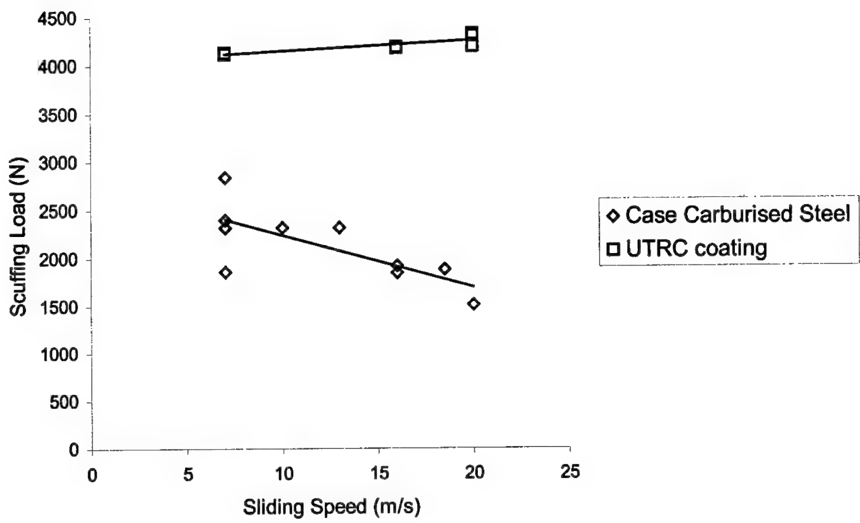

Figure 11 **Experimentally determined scuffing loads as a function of sliding velocity for the ground case carburised with optimised thin hard coating coated surface (UTRC coating) compared to reference case carburised surface tests (least square fits also shown).**

3-4-2 Test results with the superfinished surfaces

Nine tests were carried out with superfinished samples made of case carburised steel (AMS 9310) coated with the optimised thin hard coating. The results of these tests are summarised in Table 4.

Table 4 Summary of test results for superfinished case carburised steel discs coated with an optimised thin hard coating

Test n°	62	63	64	68	70	71	74	75	76
Peripheral Velocity of Fast Shaft (m/s)	9.15	9.15	9.15	20.95	20.95	20.95	26.18	26.18	26.18
Peripheral Velocity of Slow Shaft (m/s)	2.15	2.15	2.15	4.94	4.94	4.94	6.17	6.17	6.17
Mean Entraining Velocity (m/s)	5.65	5.65	5.65	12.95	12.95	12.95	16.18	16.18	16.18
Sliding Velocity (m/s)	7.00	7.00	7.00	16.00	16.00	16.00	20.00	20.00	20.00
Scuffing Load (N)	4303	4217†	4274	4176†	4304†	4322†	4295†	4323†	4314†
Maximum Bulk Temperature of Fast Disc (°C)	116	112	109	143	148	141	160	157	152
Maximum Bulk Temperature of Slow Disc (°C)	104	103	102	121	118	117	129	124	125
Mean Bulk Temperature of Discs (°C)	110	108	106	132	133	129	145	140	139
Maximum Peak Hertzian Contact Pressure (GPa)	1.72	1.71	1.72	1.70	1.72	1.72	1.72	1.72	1.72
Traction Coefficient at Scuffing Load	0.007	0.007	0.007	mal func.	0.004	0.003	0.004	0.004	0.004

†: no scuffing occurred; the load corresponds to the maximum testing load

The chart in Figure 12 gives the performance observed in terms of scuffing load versus sliding speed. Scuffing loads as shown on the graph, apart from tests n°62 and 64, correspond effectively to the maximum load as encountered since no scuffing was observed even at the highest load used in the rig. In the case of tests n°62 and 64, scuffing was observed but in both cases at the highest possible load stage. When no scuffing was observed at the maximum load the maximum contact pressure was maintained and the rig was run for a further 30 minutes.

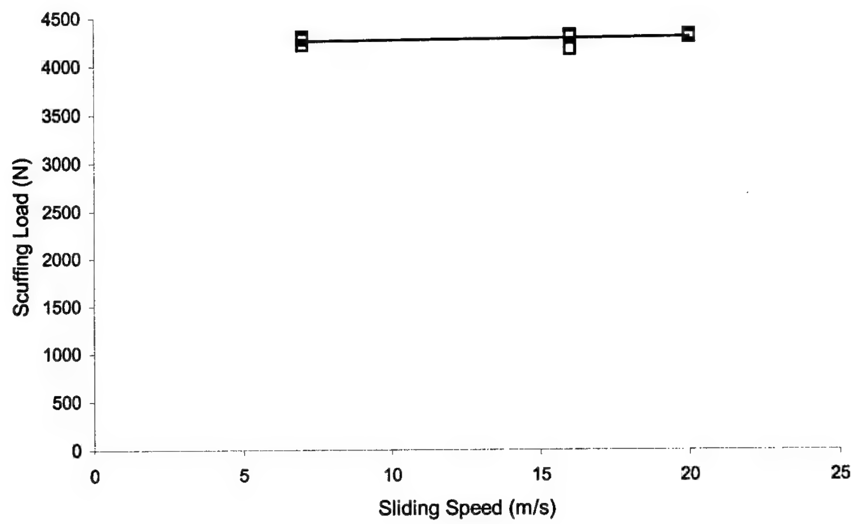

Figure 12 **Experimentally determined scuffing loads as a function of sliding velocity for the nine case carburised +superfinished+ optimised thin hard coating tests (least squares fit also shown).**

With superfinished surfaces there was little evidence of running-in as illustrated in Figure 13 which shows a surface profile of a sample before test and after testing at 16 m/s sliding speed. This behaviour is as observed with uncoated but superfinished samples where very little profile modification take place during test even in the stage preceding scuffing. Figure 14 is a surface profile of the "fast" disc of test n°62 that experienced scuffing at the highest load stage. The measurement was performed on the running track but very close to the scuffing scar. Again there is no particular evidence of any major change in the surface texture as a result of running.

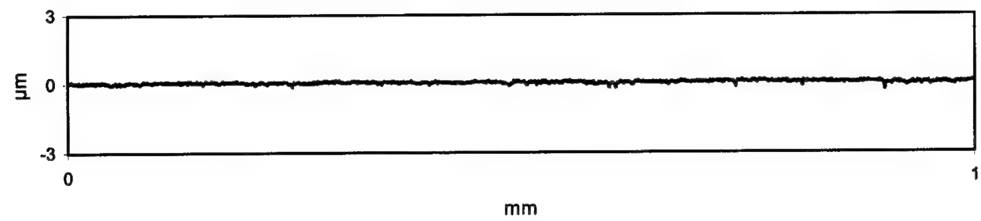

Figure 13 (a) **Surface profile from a case carburised superfinished coated disk with an optimised thin hard coating disc taken along the circumferential direction prior to testing.**

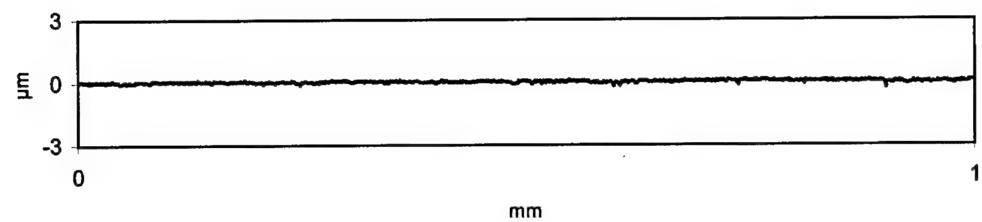

Figure 13 (b) **Surface profile from a case carburised superfinished coated disk with an optimised thin hard coating disc along the circumferential direction after test (test run at 16 m/s sliding speed).**

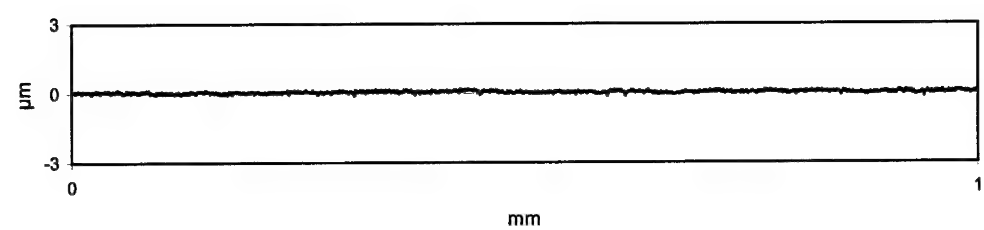

Figure 14 **Surface profile from a case carburised superfinished coated disk with an optimised thin hard coating disc along the circumferential direction, run-in but un-scuffed part of surface profile after scuffing (test run at 7 m/s sliding speed-test n°62).**

The overall scuffing performance of these samples can also be considered as remarkable particularly when compared to the baseline ground samples but also when

compared to un-coated superfinished samples. This trend is illustrated in Figure 15 where the scuffing load is plotted against sliding speed. Results on un-coated superfinished samples are from Patching et al. [6].

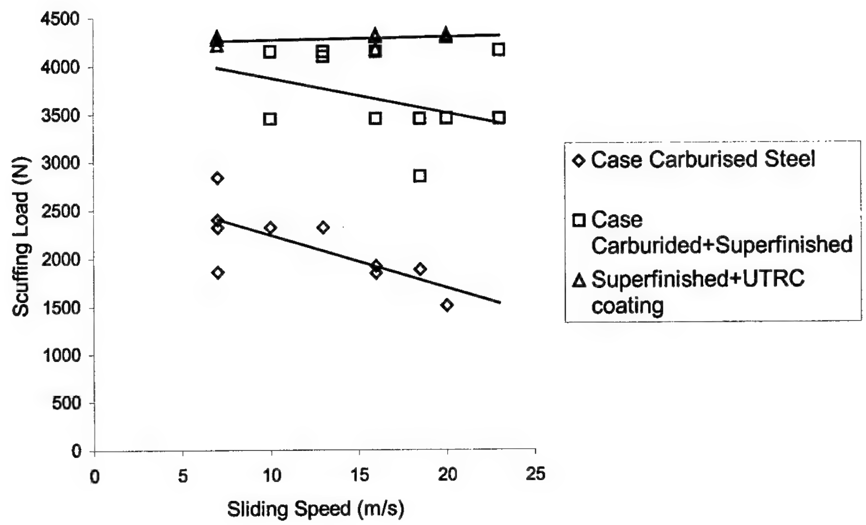

Figure 15 **Experimentally determined maximum scuffing loads as a function of sliding velocity for the superfinished case carburised with optimised thin hard coating coated surface (UTRC coating) compared to reference un-coated case carburised ground surface and superfinished surface tests (least square fits also shown).**

A striking feature of these experimental results is the relative insensitivity of scuffing to the level of sliding so that even at the highest imposed level of sliding, the coating seems to resist failure. This was not the case for un-coated superfinished surfaces where, although a major improvement of the scuffing performance was noticed compared to ground surfaces, there was a significant drop in scuffing performance at the highest sliding levels.

It is interesting to note that the two failures that did occur with the coated, superfinished samples were at the lower sliding speed of 7 m/s. This is a rather puzzling result since failure at the lower speed did not occur with the coated, ground samples. A possible explanation for this behaviour may be due to the beneficial effect (in some circumstances) of roughness in providing “fire-breaks” between micro-

contacts that effectively prevent the catastrophic spread of adhesive contact in the direction of sliding.

The appearance of the unscuffed (but run in) parts of the samples after tests was found to be consistent over the sliding speed range. As illustrated in Figure 16, the appearance does not differ significantly from the fresh, un-run parts of the samples. Figure 17 shows a scuffing scar on a superfinished sample. The scar is in the middle of the running track and the substrate metal is partially revealed. Under these severe conditions of high load (1.7 GPa) and high sliding the coating clearly reaches its protective limit. It may be noted, however, that the coating has disappeared in a fairly uniform way, and there does not appear to be any tearing of the coating on the edge of this central band. This contrasts with the behaviour as found by Alanou et al. [2] on a similar type of coating.

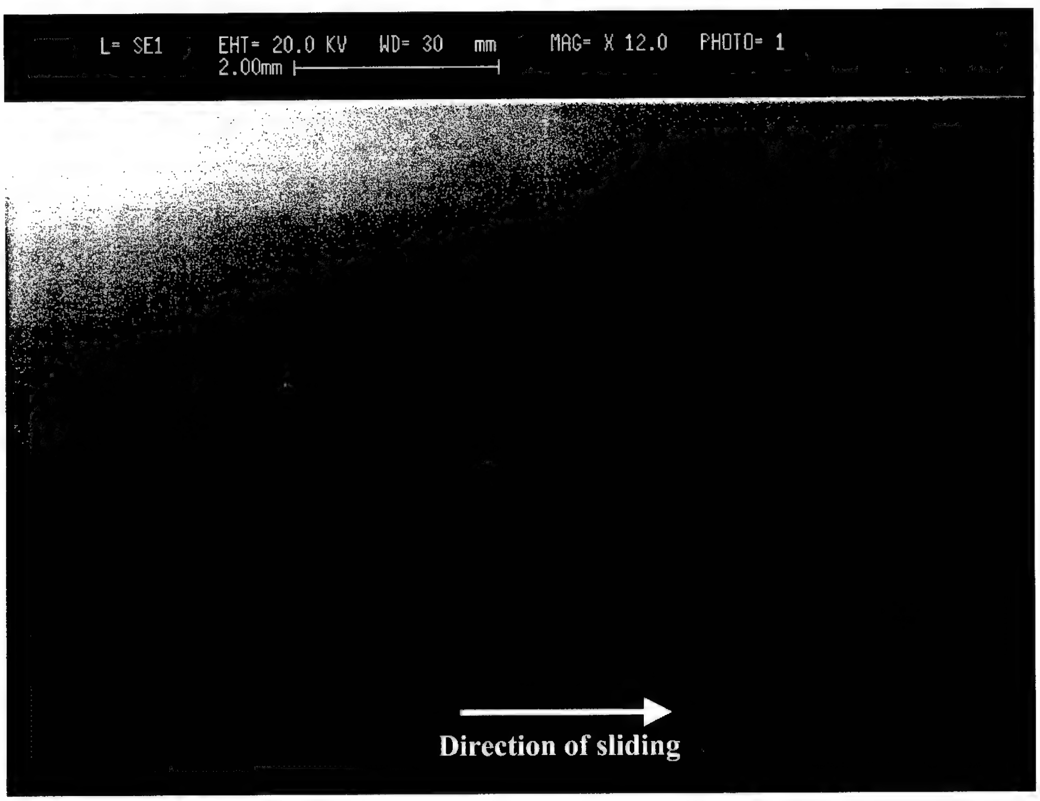

Figure 16 **SEM photograph of a superfinished coated sample after a test run at 16 m/s sliding speed, showing the slightly darker central band (clearer in reality) corresponding to the contact width.**

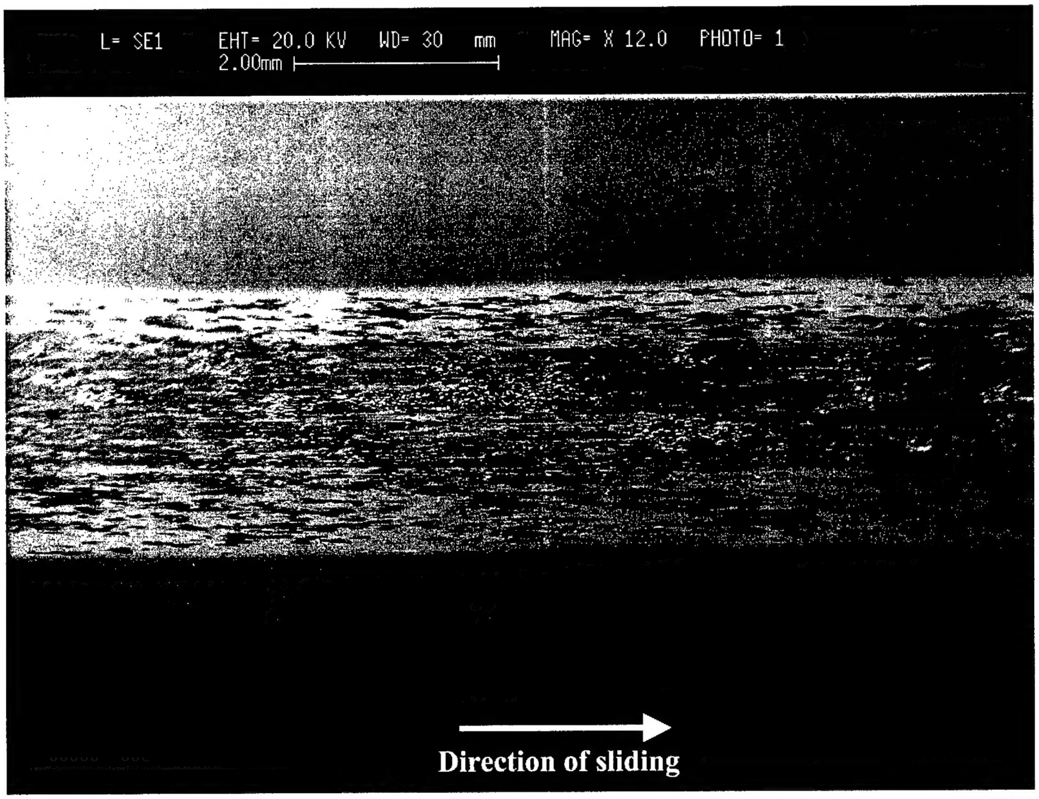

Figure 17 **SEM photograph of a superfinished coated sample after a test run at 7 m/s sliding speed, showing the scuffing scar and evidence of the bare substrate.**

Considering mean bulk temperatures of samples, the behaviour of the samples denotes a very interesting behaviour from the friction point of view. Figure 18 compares the measured mean bulk temperature of the coated/superfinished samples at the scuffing (or maximum) load with values measured in tests with ground surfaces (both coated and un-coated). The combination of coating and superfinishing is seen to give the lowest metal temperatures. The reduction in bulk temperature (compared to the reference ground/un-coated samples) is typically 100 deg C over the range of sliding speeds considered. The small degree of scatter in the results corresponding to the coated/superfinished samples may also be noted.

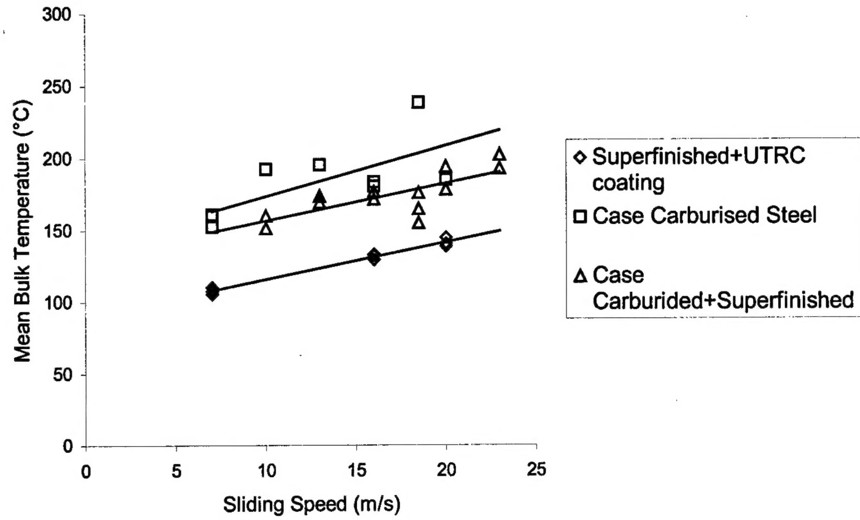

Figure 18 **Experimentally determined maximum mean bulk temperatures as a function of sliding velocity for the ground case carburised with optimised thin hard coating coated samples (UTRC coating) compared to the reference un-coated case carburised ground and superfinished samples (least square fits also shown).**

It is of interest to note in this work that superfinishing prior to coating does not appear to produce any significant additional benefit in terms of scuffing load capacity. Indeed, in these tests no scuffing was observed with the coated/ground samples, but in two of the nine tests with coated/superfinished samples scuffing occurred, albeit at the maximum load of the test rig. Superfinishing does, however, confer benefits in terms of reduced friction and lower metal temperatures.

4- Conclusions

The project was mainly concerned with the experimental determination of the relative scuffing performance of a number of different steel/surface-condition/coating combinations that may be relevant to aerospace gearing practice.

In previous work the performance of a typical case-carburised steel in both ground and superfinished state had been determined. These results served as a benchmark for the current project. The main comparisons of performance in terms of scuffing load, mean bulk temperature at scuffing, and traction coefficient are given in the body of the report. A number of interesting and potentially useful findings have emerged and the main conclusions to be drawn at this stage are as follows.

Ground/coated samples have shown the best performance; within the limits of the test rig (maximum Hertzian contact pressure of 1.7GPa) no scuffing has been experienced even at the highest load. This configuration would therefore appear to be very promising especially as a potential direct upgrade of existing designs. There are indications of incipient failure at the highest loads and speeds, however. In addition to improved scuffing capacity the presence of the coating also leads to lower friction and lower metal temperatures.

Superfinished/coated samples have (perhaps surprisingly) shown a slightly less favourable performance compared to ground/coated samples where scuffing was experienced in two instances at the lowest sliding speed. The overall performance is nonetheless very encouraging since scuffing, when it did occur, was observed only at the highest available load. In all other cases the appearance of the samples after test shows very little effect induced by the severe conditions of load and sliding speed. The scuffing that occurred in two instances seemed to have been triggered by delamination of the coating leaving the bare substrate material to resist scuffing. The superfinished/coated samples gave remarkably low metal temperatures, however, that were typically 100 deg C lower than those experienced in the tests with ground/un-coated samples.

Overall, the ultra-hard coating tested appears to be extremely promising particularly as a treatment for as-ground surfaces. It must be emphasised, however, that the endurance (fatigue) performance of the coating has not been addressed in this work and needs to be assessed separately in appropriate tests before the coating can be considered as a reliable treatment for gears.

5- References

[1] Patching, M. J., The effect of surface roughness on the micro-elastohydrodynamic lubrication and scuffing performance of aerospace gear tooth contacts", PhD Thesis, University of Wales, 1994

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